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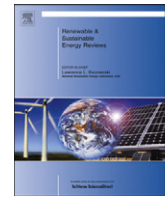

Finite-time thermodynamics optimization of absorption refrigeration systems: A review

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ABSTRACT

This paper presents a literature review of the optimization of absorption refrigeration systems based on finite-time thermodynamics. An overview of the various objective functions is presented.
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Contents

1. Introduction

The absorption refrigeration systems are thermodynamic processes which produce cold thanks to thermal energy. Then, they exchange heat with at least three sources at different temperatures without receiving work. A three-heat-source reversible refrigerator operates between heat hot reservoir, heat cold reservoir and heat sink. When T_H, T_L and T_O denote the temperatures of heat hot reservoir, heat cold reservoir and heat sink respectively, the coefficient of performance for three-heat-source reversible refrigerators is expressed as: $\varepsilon_r = [(T_H - T_O)/T_H][T_L/(T_O - T_L)]$ [1]. This expression reveals the product of thermal efficiency of Carnot cycle for heat engines working between T_H and T_O and coefficient of performance of reversible Carnot refrigerator producing cold at T_L and rejecting heat at T_O: $\varepsilon_r = \eta_C \times \varepsilon_C$ with $\eta_C = (T_H - T_O)/T_H$ and $\varepsilon_C = T_L/T_O - T_L$. In classical thermodynamics, the efficiency of a cycle operating on reversibility principles proposed by Carnot [2] became the upper bound of thermal efficiency for heat engines that work between the same temperature limits. This equally applies to the coefficient of performance of refrigeration cycles that execute a reversed Carnot cycle (Carnot refrigerator). This implies that the coefficient of performance defined above is the maximum coefficient of performance for three-heat-source refrigerators from the point of view of classical thermodynamics.

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Nomenclature

A	total heat-transfer area (m^2)
A_A	heat-transfer area of absorber (m^2)
A_C	heat-transfer area of condenser (m^2)
A_L	heat-transfer area of evaporator (m^2)
A_H	heat-transfer area of generator (m^2)
A_O	A_A+A_C
$ECOP$	ecological coefficient of performance
I	internal irreversibility parameter
K_{LC}	heat leak coefficient (W K)
K_H	thermal conductance of heat source (W K^{-1})
K_L	thermal conductance of cooled space (W K^{-1})
K_O	thermal conductance of heat sink (W K^{-1})
ncu	national currency unit
$\dot{Q}_A$	heat reject load from absorber to heat sink (W)
$\dot{Q}_C$	heat reject load from condenser to heat sink (W)
$\dot{Q}_L$	heat input load from cooled space to evaporator (W)
$\dot{Q}_H$	heat input load from heat source to generator (W)
$\dot{Q}_O$	$\dot{Q}_C+\dot{Q}_A$
R	cooling load (W)
R_m	cooling rate at maximum coefficient of performance (W)
r	specific cooling load (W m^{-2})
r_m	specific cooling rate at maximum coefficient of performance (W m^{-2})
T_1	temperature of working fluid in generator (K)
T_2	temperature of working fluid in evaporator (K)
T_3	temperature of working fluid in absorber and condenser (K)
T_3	temperature of working fluid in absorber and condenser (K)
T_A	temperature of the absorber-side heat sink (K)
T_C	temperature of the condenser-side heat sink (K)
T_{env}	temperature in environmental conditions
T_H	temperature of the heat source (K)
T_L	temperature of the cooled space (K)
T_O	$T_A=T_C$
U_H	overall heat-transfer coefficient of generator (W K/m^2)
U_L	overall heat-transfer coefficient of evaporator (W K/m^2)
U_O	overall heat-transfer coefficient of absorber and condenser (W K/m^2)
$\dot{W}$	power output (W)

Symbol

ε	coefficient of performance for absorption refrigerators
ε_C	coefficient of performance of reversible Carnot refrigerator
η_C	thermal efficiency of Carnot cycle
λ	Dissipation coefficient of cooling rate
σ	Entropy generation rate (W/K)
ε_I	coefficient of performance for three-heat-source refrigerator affected only by internal irreversibility
ε_r	coefficient of performance for reversible three-heat-source refrigerator
ε_m	coefficient of performance at maximum cooling rate

Subscripts

max	maximum

However, since the absorption refrigeration cycles are in direct contact with reservoirs and sink, the heat transfers during the isothermal processes are supposed to be carried out infinitely slowly. Therefore, duration of the processes will be infinitely long and hence it is not possible to obtain a certain amount of cooling load $\dot{Q}_L$ with heat exchangers having finite heat-transfer areas, i.e. $\dot{Q}=0$ for $0<A<\infty$. If we require certain amount of cooling load in an absorption refrigerator executing a reversible cycle, the necessary heat exchanger area would be infinitely large, i.e. $A\rightarrow\infty$ for $\dot{Q}>0$.

Thus in classical thermodynamics the real absorption refrigerators producing cold with a certain amount of cooling load are compared with the ideal absorption refrigerators developing no cooling load. In other words the performance of an absorption refrigerator of given size (in term of total heat-transfer area) is measured with an ideal absorption refrigerator which would require an infinite total heat-transfer area to produce the same amounts of cooling load. In practice, all absorption refrigeration processes take place in finite-size devices in finite-time; therefore, it is impossible to meet reversibility conditions between the absorption refrigeration system and the surroundings. For this reason, the reversible absorption cycle cannot be considered as a comparison standard for practical absorption refrigeration systems from the view of cooling load on size perspective, although it gives an upper bound for coefficient of performance. The performance bound of classical thermodynamics [3–6] is highly important in theory, but it is usually too rough to predict the coefficient of performance of practical absorption refrigerators. Therefore, it is necessary to establish the bound of finite-time thermodynamics [7].

The finite-time thermodynamics has been first proposed by Chambadal [8] and Novikov [9] independently on1957, then popularized in many works including Curzon and Ahlborn [10], De Vos [11], Sieniutycz et al. [12], Bejan [13–18], Wu [19],Chen [20], Stitou [21,22], Feidt [23,24], Leff and Teeters [25], Blanchard [26], Stitou and Feidt [27], Andresen [28], Sieniutycz and Salamon [29], De Vos [30], Bejan et al. [31], Bejan and Mamut [32], Berry et al. [33], Radcenco [34] and in many review articles including Sieniutycz and Shiner [35], Chen et al. [36], Hoffmann et al. [37] and Durmayaz et al. [38].

The finite-time thermodynamics tends to model the real systems in a way closer to reality and enable to distinguish the irreversibilities due to internal dissipation of the working fluid from those due to finite-rate heat transfer between the system and the external heat reservoirs and heat sink.

The objective of this paper is to review the present state of optimization of absorption refrigeration processes based on finite-time thermodynamics. The different performance optimization criteria are provided and discussed.

2. Optimization based on the coefficient of performance and cooling load criteria

2.1. Three-heat-source absorption refrigerator

An absorption refrigeration system (equivalent to three-heat-reservoir refrigeration system) affected by the irreversibility of finite rate heat transfer may be modeled as a combined cycle which consists of an endoreversible heat engine and an endoreversible

refrigerator that the model is shown in Fig. 1 where the temperatures of the working fluid in the Carnot engine at two isothermal processes are T_1 and T_3, the temperatures of the working fluid in the Carnot refrigerator at two isothermal processes are T_2 and T_3. T_1, T_2 and T_3 represent the temperatures of the working fluid in the endoreversible three-heat-source refrigerator when the three isothermal processes are carried out, respectively. They are respectively different from the temperatures of the corresponding three reservoirs. A single-stage absorption refrigerator consists primarily of a generator, an absorber, a condenser and an evaporator which is shown in Fig. 2. An equivalent single-stage absorption combined refrigeration system is also shown in Fig. 3. In these figures, $\dot{Q}_H$ is the heat-transfer rate from the heat source at temperature T_H to the system, $\dot{Q}_L$ is the heat-transfer rate (cooling load) from the cooled space at temperature T_L to the system, $\dot{Q}_O$ is the heat-transfer rate from the system to the heat sink at temperature T_O (in the case of three-heat-source model) and $\dot{Q}_A$ and $\dot{Q}_C$ are the heat-transfer rates from the absorber and condenser to the heat sink at temperature T_A and T_C respectively (in the case of four-heat-source model). T_1; T_2; T_3 and T_4 are the temperatures of the working fluid in the generator, evaporator, absorber and condenser. $\dot{W}$ is the power output of the heat engine which is the power input for the refrigerator. A single-stage absorption refrigerator normally transfers heat between three temperature levels when $T_A = T_C$, but very often among four temperature levels when $T_A \neq T_C$. For the three-heat-source single-stage absorption refrigerator, it is generally assumed that the working fluid in the condenser and absorber has the same temperature $T_3 = T_4$. This assumption is reasonable because the working fluid in the condenser and absorber exchanges heat with the heat sink at the same temperature. An absorption refrigerator is endoreversible absorption refrigerator when it is affected only by the external irreversibility of heat conduction between the working fluid and reservoirs.

The first performance optimization studies for absorption refrigeration systems based on finite-time thermodynamics started with the works of Yan and Chen [1,39]. They applied the theory of finite time thermodynamics in heat two-heat-source to optimize the performance of endoreversible three-heat-source refrigerator by taking the coefficient of performance and cooling load as objective functions. They treated the endoreversible three-heat-source refrigerator as a combined cycle of an endoreversible Carnot engine driving an endoreversible Carnot refrigerator as shown in Fig. 1. For convenience, they assumed that the engine and refrigerator in the combined cycle operate alternatively. Thus, the combined cycle time t supposed to be constant may be expressed as:

$$t = t_1 + t_2 \tag{1}$$

where t_1 and t_2 are the cycle times of the engine and refrigerator in the combined cycle respectively. They defined the coefficient of performance of the combined cycle as the product of the efficiency of the endoreversible Carnot engine and the coefficient of performance

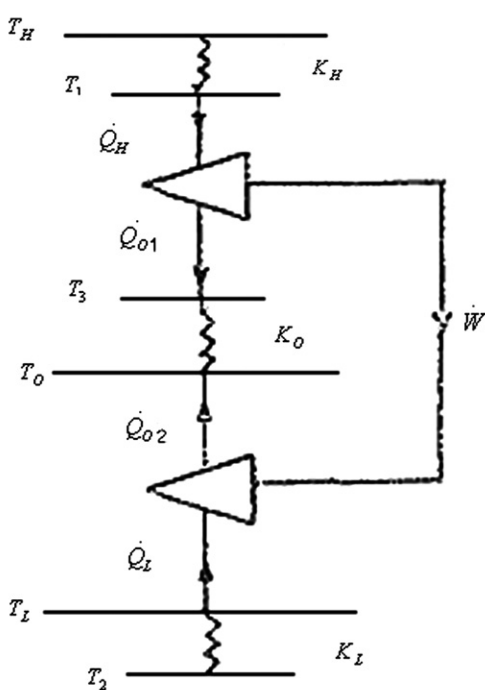

Fig. 1. Sketch of an endoreversible three-heat-source refrigerator, treated as an endoreversible Carnot engine driving an endoreversible Carnot refrigerator [1].

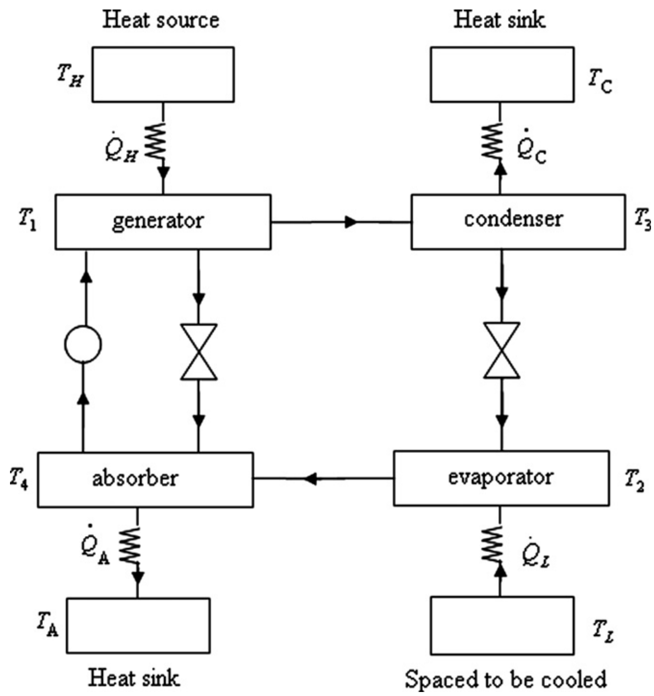

Fig. 2. Absorption refrigeration system [46].

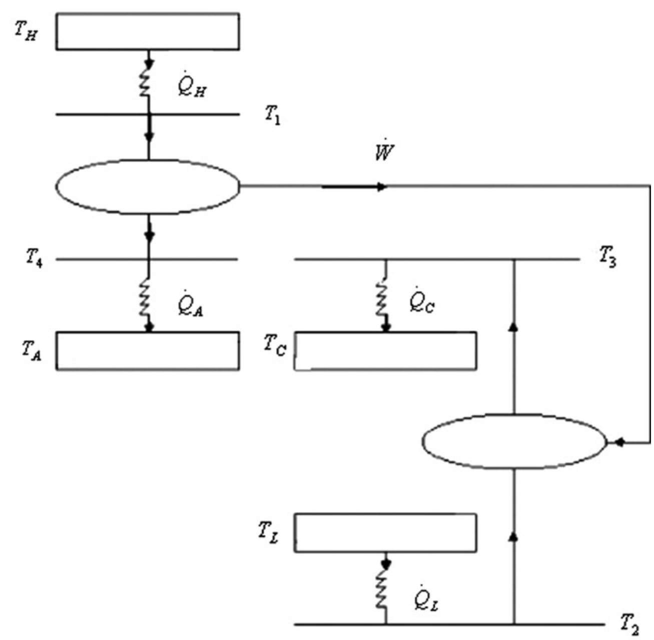

Fig. 3. Equivalent cycle of an absorption refrigeration system [46].

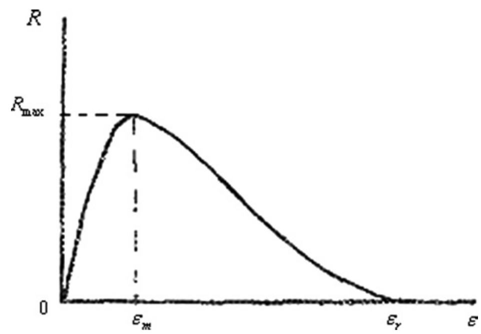

Fig. 4. $R-\varepsilon$ characteristic of an endoreversible three-heat-source refrigerator [1].

of refrigerator:

$$\varepsilon = \eta\phi \tag{2}$$

and expressed it with respect to t_1 and $t_2$taken as optimization parameters. In order to optimize ε, they introduced the Lagrangian function $L=\varepsilon+\lambda(t-t_1-t_2)$ and from the Euler–Lagrange equations $\partial L/\partial t_1$ and $\partial L/\partial t_2$ they derived the fundamental optimum relation between the coefficient of performance and cooling load of endoreversible three-heat-source refrigerator as:

$$R=\frac{K_1T_H(T_O-T_L)(\varepsilon_r-\varepsilon)}{(K_2+1)^2(1+\varepsilon)T_H-K_2{}^2T_O+(1+\varepsilon^{-1})T_L} \tag{3}$$

where $K_1=K_HK_O/(\sqrt{K_H}+\sqrt{K_O})^2$ and $K_2=\frac{\sqrt{K_O}(\sqrt{K_H}-\sqrt{K_L})}{\sqrt{K_L}(\sqrt{K_H}+\sqrt{K_O})}$

From Eq. (3), they obtained the maximum cooling load (R_{max}) and the corresponding coefficient of performance at R_{max} conditions (ε_m) shown in the following equations respectively:

$$R_{max}=\frac{K_1(\sqrt{T_H}-\sqrt{T_O})^2(T_L+F)}{T_H-T_L+G} \tag{4}$$

$$\varepsilon_m=\frac{\sqrt{T_H}-\sqrt{T_O}}{\sqrt{T_H}}\times\frac{T_L}{\sqrt{T_HT_O}-T_L+K_2\sqrt{T_O}(\sqrt{T_H}-\sqrt{T_O})} \tag{5}$$

where

$$F=\frac{K_2T_L(T_H-T_O)}{T_H-T_O}$$

$$G=K_2(\sqrt{T_H}-\sqrt{T_O})^2\left[\frac{B^2(T_H-T_O)+B\sqrt{T_H}(3\sqrt{T_H}+2\sqrt{T_O}-T_L/\sqrt{T_H})}{T_H-T_L}+\frac{3\sqrt{T_H}+\sqrt{T_O}}{\sqrt{T_H}-\sqrt{T_O}}\right] \tag{6}$$

They also obtained the behavior of the cooling rate as a function of coefficient of performance as presented in Fig. 4. Like the maximum power output and corresponding efficiency concerning a Carnot engine R_{max} and ε_m are two important performance parameters of a practical endoreversible three-heat-source refrigerator. Its optimal operating region is $\varepsilon_m \le \varepsilon \le \varepsilon_r$ [1].

Chen [40] investigated the maximum cooling rate of an endoreversible three-heat-source absorption refrigerator. He also analyzed the influence of irreversibility of finite-rate heat transfer on the performance of the system. The coefficient of performance at the maximum cooling rate was derived and the optimal performance with respect to heat transfer areas of the refrigerator was analyzed.

Bejan et al. [41] optimized the cooling load with respect to the thermal conductance of an equivalent combined endoreversible three-heat-source absorption refrigerator and obtained the analytical expression of R_{max} as:

$$R_{max}=\frac{(UA)_RT_O}{8}\left[\sqrt{\left(\frac{4W_{max}}{(UA)_RT_O}-\frac{T_L}{T_O}+1\right)^2+\frac{16T_LW_{max}}{(UA)_RT_O{}^2}}-\frac{4W_{max}}{(UA)_RT_O}+\frac{T_L}{T_O}-1\right] \tag{7}$$

and the following optimal distribution of thermal conductance inventory at the R_{max} conditions:

$$(UA)_O=\frac{UA}{2} \tag{8}$$

$$(UA)_H=(UA)_L=\frac{UA}{4} \tag{9}$$

In Eqs. (7)–(9) $W_{max}=\frac{(UA)_PT_O(\sqrt{T_H/T_O}-1)}{4}$ is the maximum power output of the endoreversible heat engine in the combined cycle; $(UA)_H$, $(UA)_L$and $(UA)_O$ are the overall thermal conductance of the respective heat exchangers; $(UA)_P$ and $(UA)_R$ are the total thermal conductance inventory of the heat and refrigerator portions; $UA=(UA)_H+(UA)_L+(UA)_O=(UA)_P+(UA)_R$ is constant and represent the total thermal conductance inventory of the entire installation. Bejan et al. [41] also reported the maximum cooling load per unit of total heat exchanger inventory.

Wijeysundera [42] focused its attention on the three-heat-source refrigerator affected only by external irreversibility of linear heat-transfer law. He investigated its optimal performance by considering the cooling load as optimization objective. He maximized the cooling capacity in term of the temperature of the heat source and the heat sink and obtained:

$$R_{max}=\beta T_H\left\{\frac{(e^2+g\phi-f-fg)+[(e^2+g\phi-f-fg)^2-4(1+g)(e^2-fg)]^{1/2}}{2(1+g)}\right\} \tag{10}$$

where $e=v+u\sqrt{\theta}$, $f=v+\theta u$, $g=v+u$ and the non-dimensional design and operating variables are defined as: $\phi=T_L/T_H$, $\theta=T_O/T_H$, $u=\gamma/\beta$, $v=\alpha/\beta$.

Wijeysundera [42] derived the following optimum relation

$$T_3/T_1=\sqrt{T_O/T_H} \tag{11}$$

and the optimal coefficient of performance at the maximum of cooling capacity condition:

$$\varepsilon_m=q_{max}(1+u\sqrt{\theta}/v)/[u(\sqrt{\theta}-\theta)-q_{max}] \tag{12}$$

where $q_{max}=R_{max}/(\beta T_H)$

The condition given by Eq. (11) is the same as that obtained by Curzon and Ahlborn for the maximum power output for an endoreversible Carnot cycle.

Wu [43,44] investigated the maximum cooling load of an endoreversible heat-engine-driven refrigerator modeled as the combined cycle of heat engine and refrigerator.

Chen and Yan [45] analyzed the effect of linear phenomenological law on the performance of three-heat-source absorption refrigerator. The heat transfer between the working fluid and the external reservoir obey the following equations:

$$Q_H=K_H(T_1{}^{-1}-T_H{}^{-1})t_H \tag{13}$$

$$Q_L=K_L(T_2{}^{-1}-T_L{}^{-1})t_L \tag{14}$$

$$Q_O=K_O(T_O{}^{-1}-T_3{}^{-1})t_O \tag{15}$$

Chen and Yan [45] defined the coefficient of performance and cooling load as the function of the temperatures T_1, T_2 and T_3 of the working fluid. By introducing the Lagrangian function $L=\varepsilon+\lambda R$ and using the Euler–Lagrange equations $\partial L/\partial x$, $\partial L/\partial y$ and $\partial L/\partial z$ where $x=T_3/T_1$, $y=T_3/T_2$ and $z=T_3$ they established the maximum cooling load, the corresponding coefficient of performance and the fundamental optimal relation between the coefficient of performance and cooling load as follows:

$$R_{max}=\frac{K_3\varepsilon_r{}^2(T_O-T_L)}{4(K_4\varepsilon_r+1)T_OT_L} \tag{16}$$

$$\varepsilon_m=\varepsilon_r/(2+K_4\varepsilon_r) \tag{17}$$

$$R=\frac{K_3\varepsilon(\varepsilon_r-\varepsilon)(T_O-T_L)}{(K_4\varepsilon+1)^2T_OT_L} \quad (18)$$

where $K_3=K_HK_L/(\sqrt{K_H}+\sqrt{K_L})^2, K_4=\frac{\sqrt{K_H}(\sqrt{K_O}+\sqrt{K_L})}{\sqrt{K_O}(\sqrt{K_H}+\sqrt{K_L})}$

Chen et al. [46] carried out similar analysis and studied the effect of the irreversibility of non linear heat transfer condition.

The performance optimization of irreversible three-heat-source refrigerator was carried out by Chen and Wu [47] to investigate the influence of irreversibility of heat leak on the performance of the Newton's law three-heat-source refrigerator. Fig. 5 shows the three-heat-source refrigeration cycle model affected by the irreversibility of finite rate heat transfer and heat leakage where Q_{LC} is the heat leak from the environment reservoir to the cooled space occurs during the full cycle time t with a thermal conductance K_{LC}. Chen and Wu [47] modeled their system by the following heat transfer Newton's law equations:

$$Q_H=K_H(T_H-T_1)t_H \quad (19)$$

$$Q_L=K_L(T_L-T_2)t_L \quad (20)$$

$$Q_O=K_O(T_3-T_O)t_O \quad (21)$$

$$Q_{LC}=K_{LC}(T_O-T_L)t \quad (22)$$

where t_H, t_L and t_O are the times of the three isothermal processes during the cycle. In such a refrigeration cycle, which consists of three isothermal processes and three adiabatic processes, the working fluid exchanges only heats with the three external heat reservoirs at temperatures T_H, T_L and T_O during the full cycle time. To give prominence to the time for heat exchange, the time for the adiabatic processes is taken as a negligible quantity because the adiabatic process is not affected by thermal resistance. The cycle time may be approximately given by $t=t_H+t_L+t_O$.

On the basic on this model, Chen and Wu [47] defined the coefficient of performance and the cooling load as:

$$\varepsilon=(Q_L-Q_{LC})/Q_H \quad (23a)$$

and

$$R=(Q_L-Q_{LC})/t \quad (23b)$$

They maximized the cooling load and the coefficient of performance of Eqs. (23a) and (23b) in term of the temperature of the working fluid in the three isothermal processes and determined analytically the maximum cooling rate and corresponding coefficient of performance, the maximum coefficient of performance and corresponding cooling rate. They derived the optimal cooling rate and the optimal coefficient of performance:

$$R=\frac{K_1(1-x)(T_Hx-T_O)T_L}{[(1+K_2)T_Hx-K_2T_O][1+K_2(1-x)]-T_Lx}-K_{LC}(T_O-T_L) \quad (24)$$

$$\varepsilon=\frac{(1-x)T_L}{(1+K_2)T_Hx-K_2T_O-T_L}\left[1-\frac{K_{LC}(T_O-T_L)}{R+K_{LC}(T_O-T_L)}\right] \quad (25)$$

where $x=T_3/T_1$, $K_1=K_HK_O/(\sqrt{K_H}+\sqrt{K_O})^2$, $K_2=\frac{\sqrt{K_O}(\sqrt{K_H}-\sqrt{K_L})}{\sqrt{K_L}(\sqrt{K_H}+\sqrt{K_O})}$

Eliminating x from Eqs. (24) and (25) yields the optimal fundamental relationship between the cooling rate and coefficient of performance. Obviously, the $R-\varepsilon$ characteristic of an absorption refrigeration cycle affected by thermal resistances and heat leak losses can be generated by the optimal fundamental relationship between the cooling rate and coefficient of performance as shown in Fig. 6. Fig. 6 presents the coefficient of performance bounds and cooling load bounds of real three-heat-source refrigerator as:

$$R_m\le R\le R_{\max} \text{ and } \varepsilon_{\max}\ge\varepsilon\ge\varepsilon_m \quad (26)$$

Jincan Chen [48] used a general irreversible cycle model to investigate the optimal performance of a class of three-heat-source affected by the three main irreversibilities which are finite rate heat transfer between the working fluid and the external heat reservoir, internal dissipation due to the working fluid and heat leakage between heat reservoirs. He established the fundamental optimal relation between the cooling rate and coefficient of performance:

$$\begin{aligned}R=&K_5\varepsilon\left[(T_H-IT_O)T_L-\varepsilon(1+C/R)T_H(IT_O-T_L)\right]\\&\times\Bigg[(1+\varepsilon)T_L+K_6^2(1+\varepsilon)\varepsilon(1+C/R)T_H\\&-K_7^2\varepsilon\left((1+C/R)/\left(1+\frac{\varepsilon}{1+\varepsilon}\frac{C}{R}\right)\right)IT_O\Bigg]^{-1}-\frac{\varepsilon C}{1+\varepsilon}\end{aligned} \quad (27)$$

where $C=Q_{LC}/t$, $K_5=\frac{K_L/I}{[1+\sqrt{K_L/IK_H}]^2}$, $K_6=\frac{1+\sqrt{K_L/IK_O}}{1+\sqrt{K_L/IK_H}}$, $K_7=\frac{\sqrt{K_L/IK_H}-\sqrt{K_L/IK_O}}{1+\sqrt{K_L/IK_H}}$

From Eq. (27) the general performance characteristic of irreversible three-heat-source refrigerator are plotted as shown in Fig. 7. Eq. (27) and the characteristic of Fig. 7 may be used directly to analyze the influence of different irreversibilities on the performance of a three-heat-source refrigerator. For example when $I=1$ and $K_{LC}=0$, the endoreversible three-heat-source affected by the irreversibility of heat transfer and heat leakage become the endoreversible three-heat-source affected only by the irreversibility of heat transfer [1]. The characteristic of Fig. 7 shows that the coefficient of performance and cooling rate should

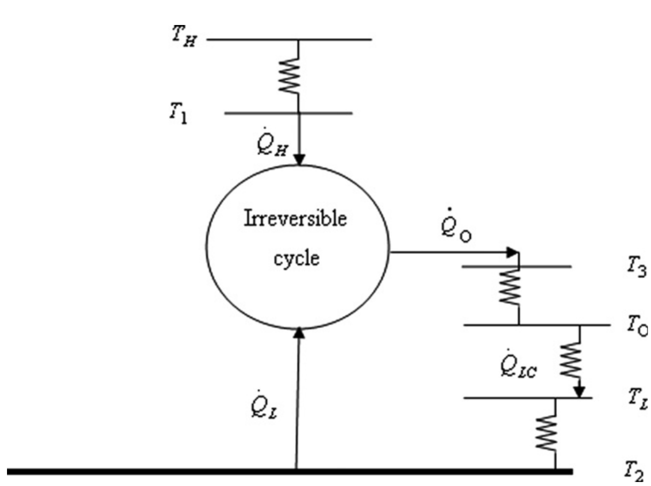

Fig. 5. A three-heat-source refrigeration cycle model [47].

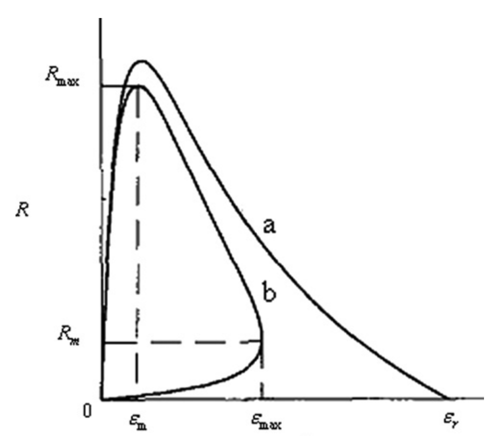

Fig. 6. $R-\varepsilon$ characteristic of a three-heat-source refrigerator affected by thermal resistances and heat leak losses. Curve a and b correspond to the cases of no heat leak loss and heat leak loss respectively [47].

be respectively, constrained by:

$$R_m \le R \le R_{\max} \text{ and } \varepsilon_{\max} \ge \varepsilon \ge \varepsilon_m \tag{28}$$

where $\varepsilon_{\max}$, $R_{\max}$, ε_m and R_m determine the upper and lower bounds of the coefficient of performance and cooling rate, respectively of the three-heat-source refrigerator affected by the irreversibility of linear heat transfer law between the working fluid and external reservoir, internal dissipation due to the working fluid and heat leakage between heat reservoirs.

An irreversible three-heat-source refrigerator affected by the irreversibility of finite rate heat transfer and internal dissipation of the working fluid was modeled as a combination of a finite size irreversible Carnot heat engine and an irreversible Carnot refrigerator [49]. The performance optimization work of this model was performed by considering the coefficient of performance as optimization criterion. The optimal overall coefficient of performance was derived and the combined effects of finite-rate heat transfer and internal dissipation on optimal performance were investigated.

The performance optimization based on the coefficient of performance and cooling load criteria for an equivalent cycle system of an endoreversible single-stage absorption refrigerator was investigated by Jincan Chen [50]. The absorption cycle and its external heat reservoir considered was modeled as an equivalent combined system which consist of a heat engine operating between heat source at temperature T_H and the heat sink at temperature T_O and a refrigerator operating between heat sink at temperature T_O and the cooled space at temperature T_L. Jincan Chen [50] considered the linear heat transfer law to model the finite-rate heat transfer between the system and its external heat reservoirs:

$$\dot{Q}_H = U_H A_H (T_H - T_1) \tag{29}$$

$$\dot{Q}_L = U_L A_L (T_L - T_2) \tag{30}$$

$$\dot{Q}_C = U_O A_C (T_3 - T_O) \tag{31}$$

$$\dot{Q}_A = U_O A_A (T_3 - T_O) \tag{32}$$

where A_H, A_C, A_A and A_L, are, respectively, the heat-transfer areas of the generator, condenser, absorber and evaporator; U_H and U_L are, respectively, the heat-transfer coefficient between the generator and evaporator and the external heat reservoir at temperatures T_H and T_L. The heat transfer coefficient between two heat exchangers, the absorber and condenser and the heat sink at temperature T_O is taken to be U_O. This is reasonable assumption because the working fluid in the absorber and condenser exchanges heat with the same heat sink at temperature T_O. The total heat-transfer areas of the heat engine and refrigerator in the combined cycle are $A_H + A_A$ and $A_C + A_L$ respectively. The total heat transfer area of the system is $A = A_H + A_A + A_C + A_L$. Chen [50] optimized the coefficient of performance and specific cooling load with respect to the total heat-transfer area of the refrigerator in the combined cycle and derived the fundamental optimum relation:

$$r = U_H \varepsilon \frac{(T_H - T_O)T_L - \varepsilon T_H (T_O - T_L)}{(1+\varepsilon)T_L - C^2 \varepsilon T_O + (1+C)^2 (1+\varepsilon)\varepsilon T_H} \tag{33}$$

where $C = \sqrt{U_H/U_r} - 1$. Eq. (33) is the same as the general optimum relation derived from the cycle model of an endoreversible three-heat-source refrigerator [40]. From the external condition $\partial r/\partial \varepsilon$ and Eq. (33), the maximum specific cooling load $r_{\max}$ and the corresponding coefficient of performance ε_m were derived. They were given by the following equations respectively:

$$r_{\max} = \frac{U_H(\sqrt{T_H} - \sqrt{T_O})^2 T_L}{T_H - T_L + 2C(T_H - \sqrt{T_H T_O}) + C^2(\sqrt{T_H} - \sqrt{T_O})^2} \tag{34}$$

$$\varepsilon_m = \frac{(1 - \sqrt{T_O/T_H})T_L}{\sqrt{T_H T_O} - T_L + C(\sqrt{T_H T_O} - T_O)} \tag{35}$$

The optimal distribution of heat-exchanger areas and the optimal temperatures of the working fluid were also determined. The $R-\varepsilon$ characteristic of the system was obtained and it is identical to that presented by Yan and Chen [1]. Jincan Chen [50] concluded that the coefficient of performance smaller than ε_m is not optimal for endoreversible three-heat-source single-stage refrigerator.

Wijeysundera [51] considered the three-heat-reservoir single-stage absorption cycles affected by internal irreversibility and discusses its specific cooling load optimization and the effect of internal irreversibilities on the performance of the system.

Chen and Schouten [52] established an irreversible single-stage absorption refrigeration cycle model which operates between three temperature levels and includes finite-rate heat transfer between the working fluid and the external heat reservoirs, heat leak from the heat sink to the cooled space, and irreversibilities due to the internal dissipations of the working fluid. This model was used to optimize the coefficient of performance and cooling rate of the system for a given total heat-transfer area of the heat exchangers with respect to the temperatures of the working fluid. Chen and Schouten [52] assumed that the heat transfer rate of the system obeys a linear law:

$$\dot{Q}_H = U_H A_H (T_H - T_1) \tag{36}$$

$$\dot{Q}_L = U_L A_L (T_L - T_2) \tag{37}$$

$$\dot{Q}_O = U_O (A_C + A_A)(T_3 - T_O) \tag{38}$$

$$\dot{Q}_{LC} = K_{LC}(T_O - T_L) \tag{39}$$

where K_{LC} is the heat leak coefficient. The irreversible cycle model mentioned above is obviously more general than an endoreversible cycle model, because it includes the major irreversibilities existing usually in real absorption refrigeration systems. Chen and Schouten [52] derived the maximum coefficient of performance ($\varepsilon_{\max}$) and corresponding cooling rate (R_m) as well as, the maximum cooling load ($R_{\max}$) and corresponding coefficient of performance (ε_m). They determined the following optimal expressions:

$$\varepsilon = \frac{(1-x)T_L}{(1+B)xT_H - BIT_O - T_L} \frac{R}{R + \dot{Q}_{LC}} \tag{40}$$

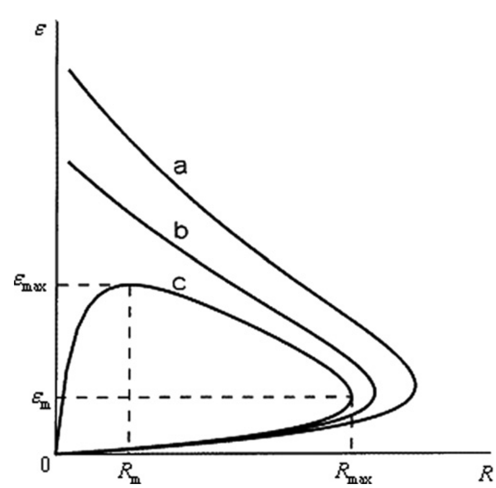

Fig. 7. $\varepsilon-R$ characteristic of a three-heat-source refrigerator affected by thermal resistances, heat leak losses and internal irreversibility. Curves a ($I=1, K_{LC}=0$), b ($I>1, K_{LC}=0$) and c ($I>1, K_{LC}>0$) are represented [48].

$$R = \frac{(1-x)(xT_H - IT_O)UT_L}{[(1+B)xT_H - BIT_O][1+B(1-x)] - xT_L} - \dot{Q}_{LC} \tag{41}$$

where $x = IT_3/T_1$, $B = \frac{\sqrt{U_O/IU_L} - \sqrt{U_O/IU_H}}{1+\sqrt{U_O/IU_H}}$ and $U = U_O/I(1+\sqrt{U_O/IU_H})^2$

They obtained the fundamental optimum relation by eliminating x from the above optimal coefficient of performance and cooling load. The optimal temperatures of the working fluid and the optimal region of temperature of the working fluid and the optimum relation for the distribution of the heat transfer areas were also calculated. From Eqs. (40) and (41) the $R-\varepsilon$ characteristic curve of the Chen and Schouten model is presented which is shown in Fig. 8. The interpretation of this figure reveals that ε_m and R_m and $\varepsilon_{\max}$ and $R_{\max}$ represent the lower bounds and the upper bounds of the coefficient of performance and cooling load respectively.

Chen et al. [53] studied the effect of heat transfer law of $q\alpha\Delta\left(T^{-1}\right)$ on the optimal performance of an irreversible three-heat-source model [52]. They considered an irreversible three-heat-source single-stage refrigerator with linear phenomenological heat transfer law affected also by the heat leak from the heat sink to the cooled space, and irreversibilities due to the internal dissipations of the working fluid. Therefore the heat transfer rates that govern the considered system are:

$$\dot{Q}_H = U_H A_H (T^{-1}{}_1 - T_H{}^{-1}) \tag{42}$$

$$\dot{Q}_L = U_L A_L (T_2^{-1} - T_L{}^{-1}) \tag{43}$$

$$\dot{Q}_O = U_O (A_C + A_A)(T_O{}^{-1} - T_3{}^{-1}) \tag{44}$$

$$\dot{Q}_{LC} = K_{LC}(T_L{}^{-1} - T_O{}^{-1}) \tag{45}$$

Applying similar optimization methods, Chen et al. [53] derived the maximum cooling load and corresponding coefficient of performance, as well as the maximum coefficient of performance and corresponding cooling rate, the optimal region of temperature of the working fluid, the optimum relation for the distribution of the heat transfer areas and the fundamental optimum relation between the coefficient of performance and cooling load. The $R-\varepsilon$ characteristic curve of an irreversible three-heat-source refrigeration model which include the irreversibility of heat transfer law of $q\alpha\Delta(T^{-1})$, heat leak from the heat sink to the cooled space and the internal dissipation of the working fluids was also derived. It is identical to that obtained by Chen and Schouten [52]. The higher and lower bounds of the coefficient of performance and cooling load with heat transfer law of $q\alpha\Delta(T^{-1})$were deduced.

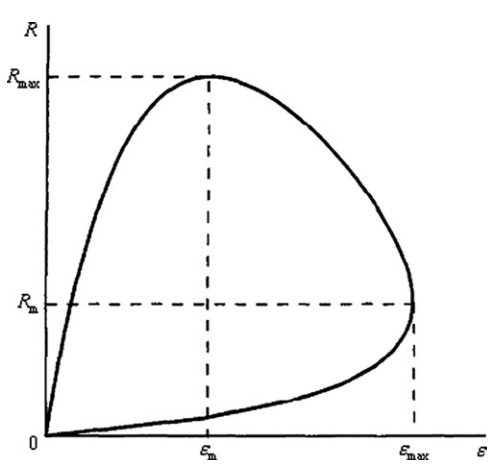

Fig. 8. $R-\varepsilon$ characteristic of a three-heat-source single-stage refrigerator affected by thermal resistances, heat leak losses and internal irreversibility [52].

Grosu et al. [54] optimized the coefficient of performance of an irreversible and endoreversible three-heat-reservoir refrigeration machine with finite time constraints on the total conductance of the system.

Baustita and Mendez [55] performed a performance optimization based on the coefficient of performance criterion of an irreversible refrigeration cycle controlled by three heat sources and affected by irreversibility of heat transfer and internal dissipation. They modeled the cycle as a combined cycle of an internally irreversible two heat source engine driving an internally irreversible two heat source refrigerator. A fundamental optimum relation between the coefficient of performance with the cooling effect was derived.

2.2. Four-heat-source absorption refrigerator

Chen [56] performed a finite time optimization analysis of a four-heat-source refrigerator affected by the heat resistance and internal dissipation inside the working fluid to determine the coefficient of performance, the temperature of the working fluid in the generator, absorber, condenser and evaporator and the useful optimal distribution relation of the heat-transfer areas at maximum specific cooling load. The maximum specific cooling ($r_{\max}$) and the corresponding coefficient of performance (ε_m) were derived. $r_{\max}$ and ε_m are two important performance parameters of absorption refrigerator operating between four temperature levels. Chen [56] determined then the upper bound for the specific cooling load and the lower bound for coefficient of performance respectively of four temperature level absorption refrigerator.

Shi and Chen [57] carried out a similar performance analysis and optimization for an irreversible four-heat-source with linear heat transfer law affected by the internal irreversibility of the working substance.

Bhardwaj et al. [58,59] investigated a performance optimization analysis based on finite-time thermodynamics approach of an equivalent endoreversible and irreversible four-heat-source absorption refrigeration system considering the thermal resistance finite thermal capacitance. In ref. [58] the optimal bounds for coefficient of performance and working fluid temperatures of the system at the maximum cooling capacity was determined. In ref. [59], they derived the maximum cooling load and the corresponding optimal coefficient of performance.

Using finite time thermodynamics, Zheng et al. [60–66] derived the fundamental optimal relation between the coefficient of performance and the cooling load, the maximum coefficient of performance and the corresponding cooling load, as well as the maximum cooling load and the corresponding coefficient of performance, the optimal working fluid temperatures, the optimal distribution relation of heat-transfer surface areas of a four-heat-reservoir affected by the irreversibilities of (i) linear phenomenological heat transfer law [62], (ii) linear phenomenological heat transfer law and heat leak [61], (iii) linear heat transfer law and heat leak [64,66], (iv) linear heat transfer law, heat leak and internal dissipation inside the working fluid [63], (v) linear phenomenological heat transfer law, heat leak and internal dissipation inside the working fluid [60,65]. They studied the effects of the cycle parameters on the coefficient of performance and the cooling load. Fig. 9 shows the four-heat-source refrigeration cycle model affected by the irreversibility of finite rate heat transfer and heat leakage. The $R-\varepsilon$ characteristic of an irreversible four-heat-source model with Newton heat transfer law and the $R-\varepsilon$ characteristic of an irreversible four-heat-source model with linear phenomenological heat transfer law obtained in Refs. [60,63] are identical to those obtained in Refs. [52,53] for an irreversible three-heat-source model with Newton heat transfer

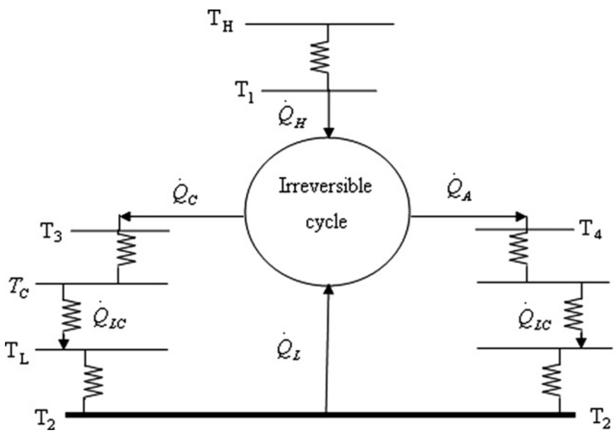

Fig. 9. A four-heat-source refrigeration cycle model [63].

law and irreversible three-heat-source model with linear phenomenological heat transfer law. These optimum characteristics may be used directly to analyze the influence of major irreversibility on the performance of an irreversible four-heat-reservoir absorption refrigeration. It can be seen from these figure that the coefficient of performance and cooling load should be constrained by:

$$R_m \leq R \leq R_{\max} \text{ and } \varepsilon_{\max} \geq \varepsilon \geq \varepsilon_m \tag{47}$$

Researches into heat-engine-driven combined vapor compression and absorption refrigerator have been conducted by Goktun [67] for space cooling to investigate their optimal performances.

2.3. Solar absorption refrigeration

Flat-plates collectors are commonly used to valorize the solar energy. The solar energy absorbed by solar collectors can be utilized to drive absorption refrigeration for cooling purposes. In a solar operated absorption refrigeration system made of a solar collector and a refrigeration cycle, the energy supplied to the generator is directly from the solar collector. The absorption refrigerator is solar absorption refrigeration when solar collector is coupled to the generator so that, the energy required by the generator is the solar energy absorbed by the solar collector. So the solar absorption refrigerator model is obtained from the model of Fig. 2 by replacing the "Heat source" by the "Solar collector" [68]. The overall coefficient of performance of the solar absorption refrigerator is equal to the product of the efficiency of the solar collector (η_s) and the coefficient of performance of the absorption refrigerator (ε_a) [68]

$$\varepsilon = \eta_s \varepsilon_a \tag{48}$$

Wu et al. [68] presented an endoreversible solar absorption refrigerator model. They derived the optimal coefficient of performance of an endoreversible solar absorption refrigerator with respect to operating temperature of the solar collector T_H:

$$\varepsilon = \frac{(1-\sqrt{T_A/T_H})T_L}{\sqrt{T_H T_A} - T_L + D\sqrt{T_A}(\sqrt{T_H} - \sqrt{T_A})} \tag{49}$$

where

$$D = \frac{\sqrt{U_A A_A / U_L A_L}(\sqrt{U_H A_H} - \sqrt{U_L A_L})}{\sqrt{U_H A_H} + \sqrt{U_A A_A}}$$

Wu et al. [68] maximized the coefficient of performance given in Eq. (49) with respect to the solar collector temperature T_H and obtained the following optimum relation:

$$(D+1)T_H{}^2 T_A{}^{1/2} + 2(T_A - T_L)T_H{}^{3/2} + T_A{}^{1/2}[(D+1)T_H$$
$$-DT_A - T_L]T_H - 2(D+1)T_A T_H{}^3 + T_H T_A{}^{1/2}(DT_A + T_L) = 0 \tag{50}$$

The resolution of Eq. (50) yields the optimum solar collector temperature. When this minimum value is substituted into Eq. (49) the optimum operating coefficient of performance for the solar endoreversible absorption refrigerator is determined.

The performance optimization research of the irreversible Carnot absorption refrigerator coupled to the corrugated sheet collector (CSC) have been conducted by Goktun and Ozkaymak [69]. The considered model is affected by the internal irreversibility in the working fluid. The optimal overall system coefficient of performance as the function of CSC temperature was [69]:

$$\varepsilon = [0.68 - 7(T_H - T_A)/I][(T_L/T_H)(RT_H - T_A)/(T_A - RT_L)] \tag{51}$$

Goktun and Ozkaymak [69] derived the optimum CSC temperature and the maximum overall coefficient of performance respectively as:

$$T_H = (T_A/\sqrt{R})\sqrt{\alpha+1} \tag{52}$$

$$\varepsilon_{\max} = [0.68 T_L R / T_A - RT_L]\{1 + (1/\alpha)[(1/R) + 1 - 2\sqrt{(\alpha+1)/R}]\} \tag{53}$$

where $\alpha = 0.68I/7T_A$ is the operating parameter of a CSC.

Fig. 10 shows the variation of the maximum overall coefficient of performance for CSC-driven irreversible Carnot absorption refrigerator with α for different value of I.

The Wijeysundera solar-powered absorption refrigeration model [70,71] is shown schematically in Fig. 11. Wijeysundera derived the maximum cooling capacity and the corresponding coefficient of performance.

Fath et al. [72] derived the maximum specific cooling load and the corresponding coefficient of performance of a four-heat-source solar absorption refrigerator.

Lin and Yan [73] and Vargas et al. [74] investigated the performance analysis and optimization of a solar-driven refrigerators. They obtained the optimal bounds of coefficient of performance and cooling capacity.

Coefficient of performance and cooling load criteria are used to evaluate the performance and the efficiency bounds of the absorption refrigerators. However, they do not give the performance limit from the view point of the thermo-economical design.

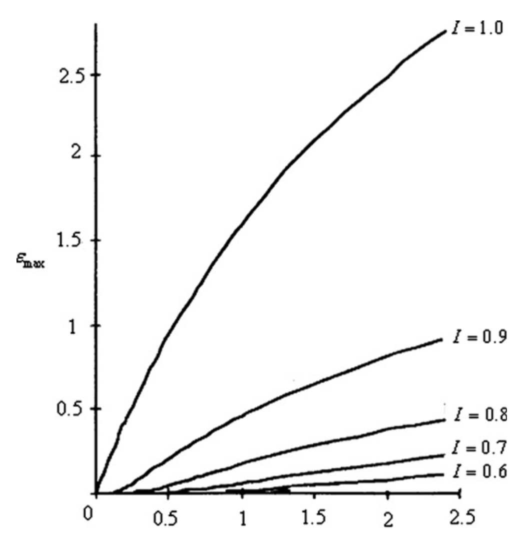

Fig. 10. The variation of $\varepsilon_{\max}$ with α for different values of I [69].

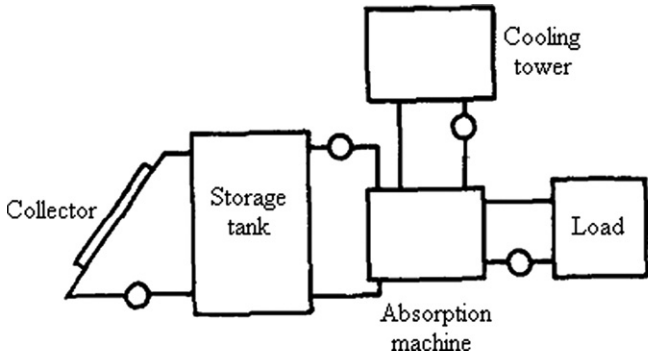

Fig. 11. Schematic diagram of solar-powered absorption system [70,71].

3. Optimization based on the thermo-economic criterion

Finite-time thermo-economic optimization is a further step in performance analysis of absorption systems based on finite-time thermodynamics to include their economic analyses. For an economical design, Sahin and Kodal [75] have introduced a new finite time thermo-economic performance criterion, defined as the cooling load for refrigerators and the heating load for heat pumps per unit total cost (total of investment and energy consumption costs). Based on this criterion, they investigated the economic design conditions of single stage and two stage vapor compression refrigerators and heat pumps [75–78]. The finite-time thermo-economic optimization technique, first introduced by Sahin and Kodal [75], has been extended to irreversible absorption refrigerators [79,80]. The objective function for an irreversible three-heat-source absorption refrigerator with not heat leak losses is defined, as [79]:

$$F = R/(C_i + C_e) \tag{54}$$

where C_i and C_e refer to annual investment and energy consumption costs, respectively. The investment cost of the absorption system is assumed to be proportional to the system size, which may be considered as the total heat transfer areas.

$$C_i = a(A_H + A_L + A_O) \tag{55}$$

where the proportionality coefficient for the investment cost of the system, a, is equal to the capital recovery factor times the investment cost per unit heat transfer area, and its dimension is ncu/(year m^2). The annual energy consumption cost is proportional to the heat rate input, i.e.

$$C_e = b\dot{Q}_H \tag{56}$$

where the coefficient b is equal to the annual operation hours times price per unit energy, and its dimension is ncu/(year kW). Substituting Eqs. (55) and (56) into Eq. (54) yields

$$F = \frac{R}{[a(A_H + A_L + A_O) + b\dot{Q}_H]} \tag{57}$$

It should be noted that the objective function defined by Eq. (57) stands as a more general form for some of the objective functions used in the literature. For example, for the special case of $a=1$ and $b=0$, the objective function becomes the specific cooling load and for $a=0$ and $b=1$, it becomes the coefficient of performance.

Kodal et al. [79] obtained the variation of the objective function for the irreversible three-heat-source absorption refrigerator with respect to the coefficient of performance for various internal irreversibility parameter I values and various economical parameter $k = a/b$ values which are shown in Figs. 12 and 13 respectively. It can be observed that the maximum thermo-economic objective function and the optimal coefficient of performance reduce as the internal irreversibility and the economical parameters increase. Moreover when $\varepsilon = \varepsilon_I = [T_L(IT_O - T_H)]/[\ T_H(T_L - IT_O)]$, the thermo-economic objective function of the three-heat-source refrigerator becomes zero. Therefore, the upper bound of the coefficient of performance given above when there is no finite rate heat transfer irreversibility does not have very much instructive significance for practical applications.

Kodal et al. [79] maximized the objective function with respect to the working fluid temperature and derived the optimum working fluid temperature, the optimum coefficient of performance and the optimum specific cooling load. The optimal distribution of the heat exchangers areas are also obtained for a given total heat transfer area (i.e. $A = A_H + A_L + A_O$). The effects of the internal irreversibility, the economic parameter ($k = a/b$) and the external temperatures on the global and optimal economic performances were discussed.

Qin et al. [80] analyzed and optimized the thermo-economic performance of an endoreversible four-heat-reservoir absorption

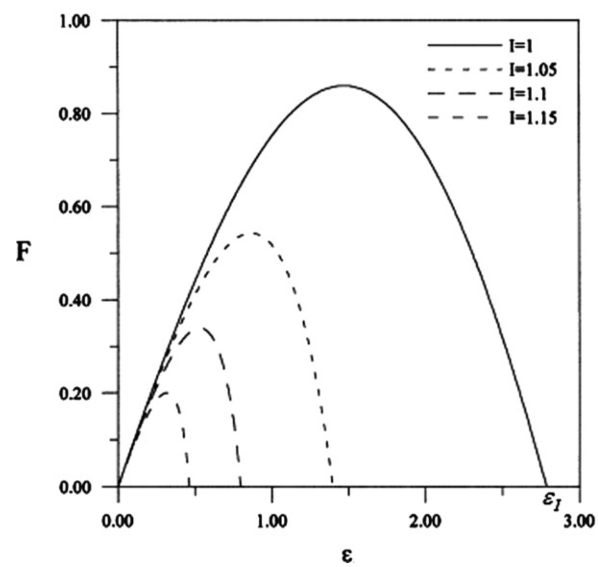

Fig. 12. Variation of the thermoeconomic objective function for three-heat-source refrigerator with respect to the coefficient of performance, for various I values [79].

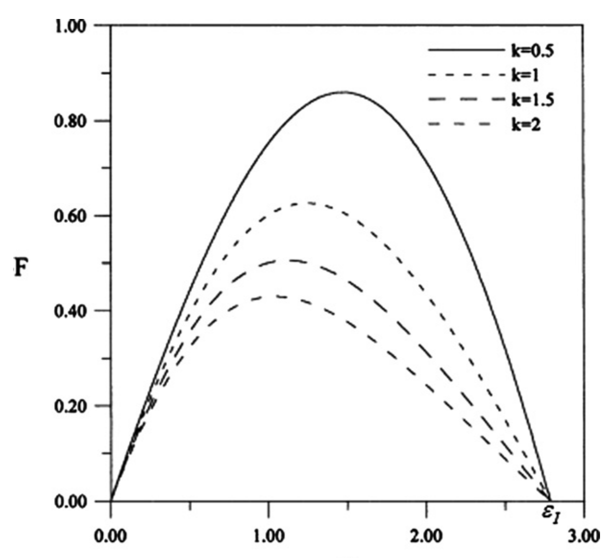

Fig. 13. Variation of the thermoeconomic objective function for three-heat-source refrigerator with respect to the coefficient of performance, for various k values [79].

refrigeration cycle assuming a linear heat transfer law. In this study the total heat transfer surface area of the four-heat-exchangers is assumed to be constant. The optimal relation between the thermo-economic criterion and the coefficient of performance, the maximum thermo-economic criterion, and the corresponding coefficient of performance (ε_F) and specific cooling-load (r_F) were derived. They derived the optimal relation between the thermo-economic criterion and the coefficient of performance of an endoreversible absorption refrigerator.

Applying this optimal relation, the thermo-economic performance characteristics of an endoreversible four-heat-source absorption refrigerator with a linear heat transfer law can be derived. Figs. 14 and 15 show the $F-\varepsilon$ characteristic curve and the $F-r$ characteristic curve respectively of the endoreversible absorption refrigerator. According to these figures the optimal region of the endoreversible absorption refrigerator should obey the following relations:

$$F_r \le F \le F_{\max}, \varepsilon_F \le \varepsilon \le \varepsilon_r, r_F \le r \le r_{\max} \tag{58}$$

where F_r is the thermo-economic criterion for the maximum specific cooling load ($r_{\max}$), and ε_r is the coefficient of performance for the maximum specific cooling load ($r_{\max}$) of the endoreversible absorption refrigerator. Qin et al. [80] also provided numerical examples to discuss the effects of the cycle parameters on the characteristic of the cycle.

The thermo-economical objective function F is used to reduce as well as possible the costs in the design and the industrial facility of the absorption refrigerators and then to make the savings in their thermal consumption of energy. However, like the coefficient of performance and cooling load criteria, he only

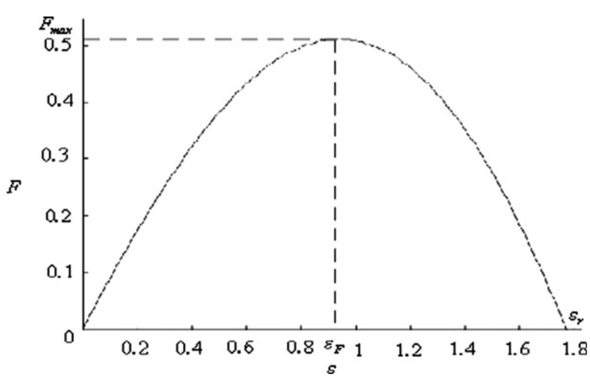

Fig. 14. Variation of the thermoeconomic objective function for an endoreversible four-heat-source refrigerator with respect to the coefficient of performance [80].

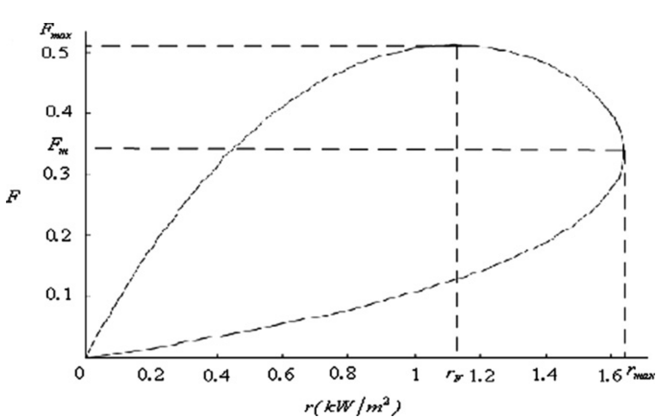

Fig. 15. Variation of the thermoeconomic objective function for an endoreversible four-heat-source refrigerator with respect to the specific cooling load [80].

takes into account the first law of thermodynamics and therefore he does not describe the performance of the absorption refrigerators from the view point of the inevitable degradations of energy which occur in the system during the refrigerating cycle of the working fluid. This aspect is taken into account by the second law of thermodynamics and appears in the thermo-ecological criteria.

4. Optimization based on the ecological criterion

Angulu-Brown [81] proposed an ecological optimization function E which is expressed as:

$$E = \dot{W} - T_L\sigma \tag{59}$$

where $\dot{W}$ is the power output and σ is the entropy generation rate. Yan [82] discussed the results of Angulu-Brown [81] and suggested that it may be more reasonable to use:

$$E = \dot{W} - T_O\sigma \tag{60}$$

if the cold reservoir temperature is not equal to the environment temperature T_O. The optimization of ecological function is therefore claimed to achieve the best compromise between the work-energy rate (e.g. power of an engine, cooling rate of a refrigerator, or heating rate of a heat pump) and its dissipation which is produced by entropy generation in the system and its surroundings. Chen et al. [83] and Yan et Lin [84] reported a similar ecological optimization of the three-heat-source absorption refrigerator. They defined the ecological criterion function E of a refrigerator as:

$$E = R - \lambda T_O\sigma \tag{61}$$

where λ is the dissipation coefficient of the cooling rate. The physical meaning of the dissipation coefficient of cooling rate λ is that, in theory, if the rate of availability $T_O\sigma$ were not lost, it would produce a cooling rate $\lambda T_O\sigma$ through a reversible Carnot refrigerator operating between T_O and T_L. This shows that λ is equal to the coefficient of performance of the reversible Carnot refrigerator, i.e.

$$\lambda = T_L/(T_O - T_L) \tag{62}$$

In their performance ecological optimization work for an irreversible three-heat-source absorption refrigerator with linear heat transfer law affected by internal dissipation inside the working fluid, Yan et Lin [84] investigated the maximum ecological optimization criterion E and derived the optimal cooling load, coefficient of performance and entropy production rate i.e.

$$\varepsilon_E = \frac{\varepsilon_r\varepsilon_I T_H/T_L - 2 + [(\varepsilon_I T_H/T_L + 1)(\varepsilon_r T_H/T_L + 2)(\varepsilon_I + 1)(\varepsilon_r + 2)]^{1/2}}{(\varepsilon_r + 2\varepsilon_I)T_H/T_L + 2(T_H/T_L + 1)} \tag{63}$$

$$R_E = \frac{\alpha A}{(1+\sqrt{I})^2}\left(\frac{T_H}{T_H/T_L + \varepsilon_E{}^{-1}} - \frac{IT_O}{1+\varepsilon_E{}^{-1}}\right) \tag{64}$$

$$\sigma_E = \frac{\alpha A(1 - T_O/T_H)(\varepsilon_E^{-1} - \varepsilon_r^{-1})}{T_O(1+\sqrt{I})^2}\left(\frac{T_H}{T_H/T_L + \varepsilon_E^{-1}} - \frac{IT_O}{1+\varepsilon_E^{-1}}\right) \tag{65}$$

where $\varepsilon_I = (1 - IT_O/T_H)T_L/(IT_O - T_L)$ is the upper bound of coefficient of performance for the irreversible three-heat-source absorption refrigerator.

ε_E, R_E and σ_E are three important parameters of an irreversible three-heat-source refrigerator, they determine the ecological optimization performance of the refrigerator. The optimal temperature of the working fluid in the three isothermal processes were also obtained. By numerical example they showed some advantages in using the ecological optimization criterion; it is

especially beneficial for determining the reasonable use of energy and protecting the ecological environment. The ratios $R_E/R_{\max}$, σ_E/σ_m and $(\sigma_E/\sigma_m)/(R_E/R_{\max})$ against I curves are shown in Fig. 16. It can be observed from this figure that the principal advantage of the ecological optimization criterion over the maximum work-energy rate criterion is a great reduction in the ratio between entropy production and work-energy rate for the refrigerator.

Huishan [85] carried out similar ecological performance analysis to determine the influence of finite time heat transfer between the heat sources and the working fluid on the ecological optimal performance of a three-heat-source absorption refrigerator.

Chen et al. [86,87] also reported similar performance characteristic of an irreversible absorption refrigeration system at maximum ecological criterion operating among four temperature levels. They derived the optimal relationship between the ecological criterion and the coefficient of performance.

The thermo-ecological criterion is used to achieve the best compromise between the cooling rate and its dissipations of the absorption refrigerators. However, it may take negative values. Such an objective function in a performance analysis can be defined mathematically; however, it needs interpretation to comprehend this situation thermodynamically.

5. Optimization based on the new thermo-ecological criterion

Ust [88] has recently introduced a new dimensionless ecological optimization criterion called the ecological coefficient of performance (ECOP) that has always positive values and takes into account the loss rate of availability on the performance. Ust [88] defined the *ECOP* as the ratio of power output to the loss rate of availability:

$$ECOP = \frac{\dot{W}}{T_{env}\dot{\sigma}} \tag{66}$$

By employing the ECOP function, many studies have been done for different heat engine models [89–94]. The ECOP function defined for heat engines has been modified for irreversible Carnot refrigerator model by Ust and Sahin [95,96], as the ratio of cooling load to the loss rate of availability:

$$ECOP = \frac{\dot{R}}{T_{env}\dot{\sigma}} \tag{67}$$

The ECOP give information about to the loss rate of availability or entropy generation rate in order to produce a certain cooling. It should be noted that for a certain cooling load, the entropy generation rate is minimum at maximum ECOP condition. The maximum of the ECOP function signifies the importance of getting the cooling load from a refrigerator by causing lesser dissipation in the environment. Therefore the higher the ECOP, we have a better absorption refrigerator in terms of cooling load and the environment considered together.

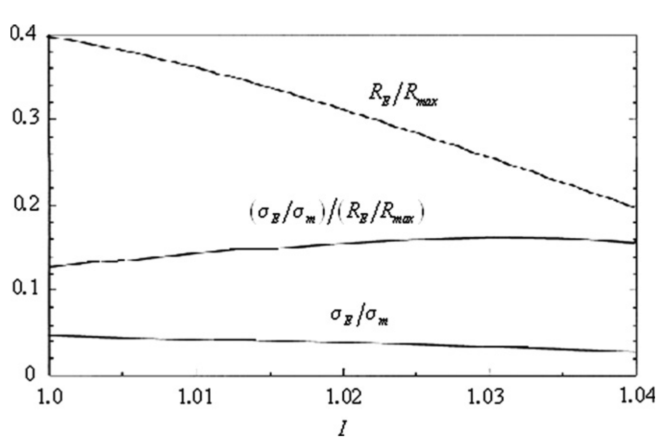

Fig. 16. The $R_E/R_{\max}$, σ_E/σ_m and $(\sigma_E/\sigma_m)/(R_E/R_{\max})$ against I curves of the irreversible three-heat-source refrigerator [84].

Very recently Ngouateu Wouagfack and Tchinda [97] extended a thermo-ecological performance analysis based on ECOP criterion given in Refs. [95,96] to an irreversible three-heat-source absorption refrigerator which includes finite-rate heat transfer between the working fluid and the external heat reservoirs, heat leak from the heat sink to the cooled space, and irreversibilities due to the internal dissipations of the working fluid. They determined analytically the maximum of the ecological performance criterion and the corresponding optimal coefficient of performance, cooling load and entropy generation rate for a given total heat-transfer area of the heat exchangers. The corresponding optimal temperatures of the working fluid in the main components of the system and the optimal distribution of the heat-transfer areas are also obtained analytically. The influences of the major irreversibilities on the thermo-ecological performances are discussed detailed. Additionally, the variations of the normalized ECOP and COP with respect to the entropy generation rate have been demonstrated which is shown in Fig. 17. From this figure and analytically, Ngouateu Wouagfack and Tchinda [97] obtained that the maximum of the ECOP and COP coincides.

Ngouateu Wouagfack and Tchinda [98] considered an irreversible three-heat-source absorption refrigeration model with the linear phenomenological heat transfer law of $\dot{Q}\alpha\Delta(T^{-1})$ which includes heat leak from the heat sink to the cooled space, and irreversibilities due to the internal dissipations of the working fluid. They optimized the specific cooling load, the COP, the ecological function E and the ECOP and established that the absorption refrigeration cycle working at maximum ECOP conditions has a significant advantage in terms of entropy production rate and coefficient of performance over that working at maximum E or maximum R conditions.

6. Conclusion

Finite-time thermodynamics optimization is the optimization method of various real thermodynamic processes and devices affected by the irreversibility of heat transfer with their surroundings. It is different from classical thermodynamics optimization which does not take into account the thermal resistances.

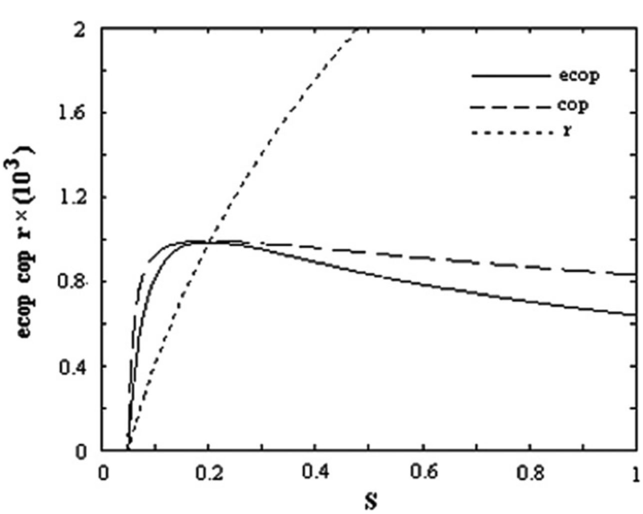

Fig. 17. Variation of the normalized ECOP, COP and the specific cooling load with respect to the specific entropy generation rate [97].

In this paper an overview of the performance optimization criteria based on the finite-time thermodynamics for absorption refrigerator systems has been presented. The coefficient of performance, the cooling load, the thermo-economic objective function, the thermo-ecological objective function and the new thermo-ecological objective function have been discussed. It has been seen that the major irreversibilities such as thermal resistance, heat leak and internal irreversibilities due to the dissipation of the working fluid affect the performance of real absorption refrigeration systems. The coefficient of performance, the cooling load, the thermo-economic criteria take into account only the first law of thermodynamics and therefore, they do not describe the performance of the absorption refrigeration cycles from the view point of the thermo-ecological aspect. This factor is taken into account by the second law of thermodynamics characterized by the entropy production which appears in the ecological optimization criterion (E) and the ecological coefficient of performance (ECOP). With the requirement of a rigorous management of our energy resources, one should have brought to be interested more and more in the second principle of thermodynamics, because degradations of energy, in other words the entropy productions, are equivalent to consumption of energy resources. The ecological function E criterion can take negative values. At this condition, the loss of cooling load is greater than the cooling load produced. The ECOP criterion is dimensionless and has always positive values.

This literature review is a contribution for the development of real absorption refrigeration systems since it may provide a general theoretical tool for their design. Moreover, it is hoped that this contribution will stimulate wider interest in the definition of new performance criteria for the optimization of absorption refrigerators.

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